Applied Thermal Engineering 143 (2018) 660–675

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

A recent review of waste heat recovery by Organic Rankine Cycle

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HIGHLIGHTS

- Studies on ORC for waste heat recovery during the last four years are reviewed.
- Reviews are on cycle configuration, working fluids, and operating condition.
- Mostly studied heat sources have been internal combustion engines and gas turbines.
- Statistics on fluids and configurations are reported for mostly studied sources.

ARTICLE INFO

Keywords:
Organic Rankine cycle
Waste heat
Working fluid
Cycle efficiency

ABSTRACT

The increment of using fossil fuels has caused many perilous environmental problems such as acid precipitation, global climate change and air pollution. More than 50% of the energy that is used in the world is wasted as heat. Recovering the wasted heat could increase the system efficiency and lead to lower fuel consumption and CO2 production. Organic Rankine cycle (ORC) which is a reliable technology to efficiently convert low and medium temperature heat sources into electricity, has been known as a promising solution to recover the waste heat. There are numerous studies about ORC technology in a wide range of application and condition. The main objective of this paper is to presents a review of studies both theoretical and experimental on ORC usage for waste heat recovery and investigation on the effect of cycle configuration, working fluid selection and operating condition on the system performance, that have been developed during the last four years. Finally, the related statistics are reported and compared regarding the configuration and the employed working fluid with type of the heat source.

1. Introduction

During the last decades, human dependency on energy has increased [1,2]. Recent global oil consumption was about 76 million barrels per day [3]. This situation can lead to various challenges such as acid precipitation [4], ozone layer depletion [5], and global climate change and air pollution [6], while the global energy resources continue to decrease [7]. There were two main approaches for overcoming the environmental problems; the first is to develop and enhance the use of renewable energy sources like solar, wind [8], biomass [9], and geothermal [10]. The second approach is to find a way for enhancement of energy conversion systems so that the system efficiently uses the energy that can be received from a source [11–13]. The evidences show that more than 50% of the energy that is used in the world is wasted as heat [1,14]. There are various energy systems with high level of waste heat such as gas and steam turbines, internal combustion engines, industrial and household waste heat, as well as geothermal heat, biomass heat, and solar radiation as shown in Fig. 1 which is statistics of the type of heat sources of the recent investigations reviewed in this study [15–17].

Using the available technologies for power generation from waste heat increases the system efficiency which can lead to fuel consumption reduction [18]. For instance, it is shown that waste heat recovery in a midsize cement plant can enhance the energy efficiency up to 20% and reduce the CO2 pollution up to 10,000 tons/year [19]. The parameters that can affect the feasibility of waste heat recovery are flow rate, temperature, and enthalpy or specific heat of the waste heat stream [20]. Generally, there are four categories of waste heat energy which are liquid streams, flue gases, steam and process gases and vapors. The temperature range for each category is 50–300 °C, 150–800 °C, 100–250 °C and 80–500 °C, respectively [21]. There are some advantages for waste heat among renewable resources. Unlike the wind or hydro sources, waste heat does not require development of land or extraction of resources. Compared to solar or wind energies, waste heat

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https://doi.org/10.1016/j.applthermaleng.2018.07.136
Received 6 January 2018; Received in revised form 25 July 2018; Accepted 29 July 2018
Available online 30 July 2018
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Nomenclature

HT	high temperature
LT	low temperature
HTL	high temperature loop
LTL	low temperature loop
CCE	confluent cascade expansion
EGMR	exhaust gas mixture recirculation
ORC	Organic Rankine cycle
IHE	internal heat exchanger
RIT	radial inflow turbine
EPC	electricity production cost
EG	exhaust gas
EGR	recirculation exhaust gas
JW	jacket water
CA	charge air
DL	dual loop
SL	single loop
DPP	depreciated payback period
KC	Kalina cycle
VFR	volume flow ratio
EDF	exergy destruction factor
SP	turbine size parameter
WHR	waste heat recovery
OS	thermal-oil storage
DE	diesel engine
RC	steam Rankine cycle
TRC	transcritical Organic Rankine cycle
LNG	liquefied natural gas
BOG	boil-off gas
D4	octamethylcyclotetrasiloxane
MDM	octamethyltrisiloxane
MD2M	decamethyltetrasiloxane
HST	heat source temperature
IHE	internal heat exchanger
BC	Brayton cycle
TLC	trilateral power cycle
ARC	Absorption Refrigeration Cycle

has high potential utilization factor [20,22,23]. There are various technologies using waste heat like steam Rankine cycle, Kalina cycle, Tilateral flash cycle, supercritical CO2 cycle, Brayton cycle, Stirling cycle and organic Rankine cycle (ORC) [24–27]. The mentioned cycles have following difficulties compared to an ORC. Organic Rankine cycle (ORC) is a well-known promising solution to recover waste heat because of its advantages such as flexibility, high safety, low maintenance requirements and good thermal performance [13,28,29]. The ORC system is similar to the conventional Rankine cycle; however it uses refrigerants or volatile organic liquids as the working fluid instead of water [30–32]. Organic working fluids because of the lower boiling points compared to water, make it possible to recover energy from low temperature waste heat sources [33]. The thermal efficiency of an ORC system depends on the thermodynamic properties of the working fluid and operating conditions of heat source, sink and cycle [34]. Generally, the range of average thermal efficiency of an ORC system is from 0.02 to 0.19, and for small systems (lower than 5 kW) has lower thermal efficiency [35–37]. In this paper, we have studied the recent reported researches on the ORC systems which utilize waste heat as heat source to produce power.

2. System configuration and components

The major components of an ORC are evaporator, condenser, expander, pump, working fluid and waste heat source [38]. In this section, a quick review of different types and components are presented.

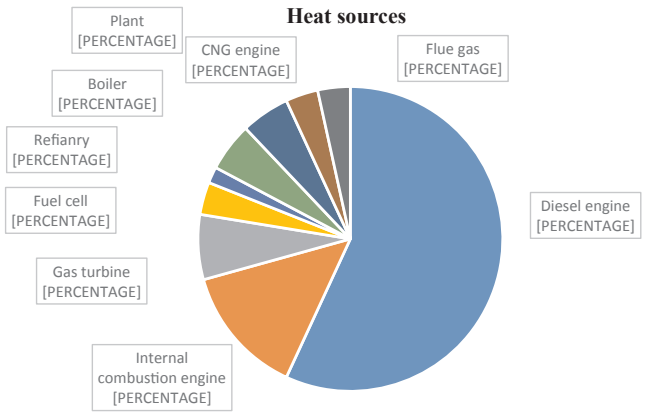

Fig. 1. Different heat sources for waste heat recovery.

2.1. Types of ORC

2.1.1. Basic ORC (BORC)

BORC works in subcritical conditions and requires the minor number of components in comparison to the other types of ORC. As can be seen in Fig. 2, BORC has four different processes; isentropic compression (3-4), heat addition (4-1), isentropic expansion (1-2), and heat rejection (2-3) [39–42].

2.1.2. Single stage regenerative ORC (SRORC)

Fig. 3 shows a SRORC system. In this system, a portion of the vapor is taken out between two stages of the turbine and added into the feedwater heater [41–43]. The regenerator can enhance the cycle efficiency by reducing the addition of heat from the evaporator heat source [44].

2.1.3. Double stage regenerative ORC (DRORC)

Fig. 4 shows a DRORC system. This system is also like SRORC but the extraction is happened between two stages. The DRORC enhances the cycle thermal efficiency by decreasing the evaporator load [42–44].

2.1.4. Reheat ORC (RORC)

Fig. 5 shows a RORC system. In this system, vapor from the evaporator section at high pressure enters into the first turbine. Then the outlet vapor re-enters to the evaporator and is reheated using the heat source before entering to the second lower pressure turbine. The purpose of RORC system is to eliminate the moisture of the steam at the final stages of the expansion process [44].

2.1.5. ORC with recuperator

Fig. 6 shows an ORC with a recuperator. In this system in order to enhance the efficiency, the high temperature working fluid from the turbine flows through the low pressure side of IHX and low temperature working fluid from the pump flows through the high pressure side of IHX [44–47].

2.1.6. Dual loop ORC (DLORC)

Fig. 7 shows a DLORC system. In this system, the HT loop is used to recover the waste heat source. The LT loop is used to recover the jacket cooling water and the excess heat of the HT loop. This system enhances the overall efficiency of the cycle by decreasing the heat load that dissipated to the environment [48–50].

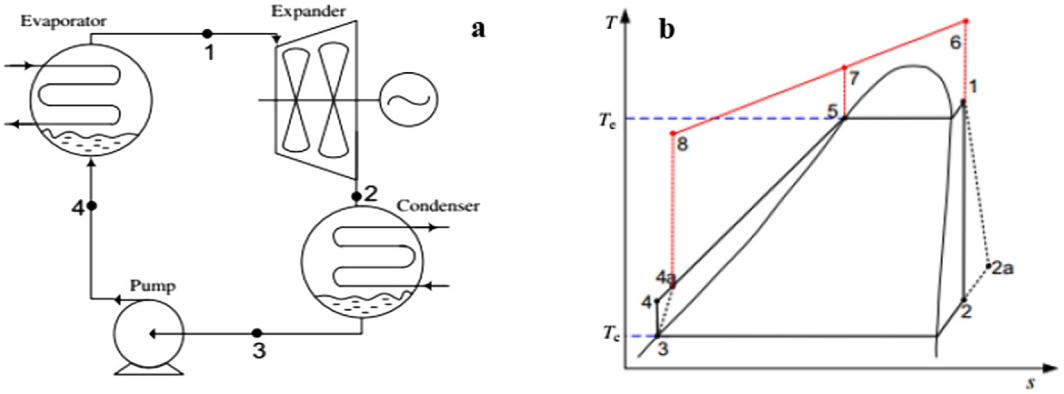

Fig. 2. (a) Schematic of basic ORC (BORC) and (b) T-S diagram for BORC [42].

2.2. Components

Working fluid, expander, heat exchanger, pump and waste heat source are the major components in an ORC system.

2.2.1. Expanders

Generally, there exist two types of expanders: one is the velocity type, such as axial and radial flow turbines, and the other is volume type, such as screw, scroll, reciprocal piston expanders and rotary vane expanders [51]. There are three types of turbines: axial flow, radial inflow (RIT) and radial outflow turbine (ROT). RIT requires smaller and fewer stages compared to the axial flow turbine that decreases the cost and improves the compactness. ROT has higher efficiency than an axial turbine [52–55]. Usually, scroll expander combines two spiral wraps in which one is fixed and the other rotates. There are two types of scroll expander: compliant and kinematically constrained [56,57]. Screw expanders include a pair of meshing helical rotors, a male and a female rotor, contained in a casing which surrounds them with clearances [51]. Reciprocating piston expanders are complicate devices which need accurate inlet and exhaust valve timing. The most advantage of reciprocating piston expanders is that they can tolerate a wet expansion [58]. Rotary vane expanders can tolerate a wide range of vapor qualities of the working fluid [59,60].

2.2.2. Working fluids

Generally, organic fluids are heavy compounds with large molecular weights and low boiling temperatures and pressures [61,62]. One of the most important ways to characterize the organic fluids is using the slope of their saturation vapor curve as shown in Fig. 8. Dry, wet and isentropic fluids have positive, negative and infinite slopes, respectively [63,64]. For an ORC with lower operating temperatures, dry and isentropic fluids show better performances compared to the wet fluids. For ORC systems with low-grade waste heat sources, organic fluids with lower latent heat of vaporization show better thermal performances [64]. Organic fluids with low specific volumes lead to smaller heat exchanger and expander sizes, reducing the size and cost of the system noticeably [65]. It is better for cycle efficiency to have the critical temperature of the organic fluid close to the maximum temperature of the heat source [66]. The freezing point of the organic fluid must be lower than the lowest temperature of the cycle. Higher molecular weight, lowers the number of stages required for the expander that reduces the cost and complexity [67].

3. Waste heat recovery of organic Rankine cycles (ORCs)

Generally, the investigations that have been done in recent years (since 2014) for improvement of the system performance can be categorized into three sections: effect of cycle configurations, working fluid selection and operational working conditions.

3.1. Effect of configurations on waste heat recovery

Nazari et al. [68] presented subcritical steam Rankine cycle that was combined with a transcritical organic Rankine cycle for waste heat recovery from a gas turbine as shown in Fig. 9. They performed parametric study in order to investigate the effects of important parameters such as: organic turbine inlet pressure, organic preheater pinch temperature and organic condensation temperature. Similarly, Shu et al. [69] investigated a transcritical cascade-ORC for multi-grade waste heat recovery from heavy-duty diesel engine. The maximum heat recovery from EG, EGR, CA and JW were 153.0 kW, 9.1 kW, 37.5 kW and 267.0 kW, respectively, which showed remarkable heat recovery

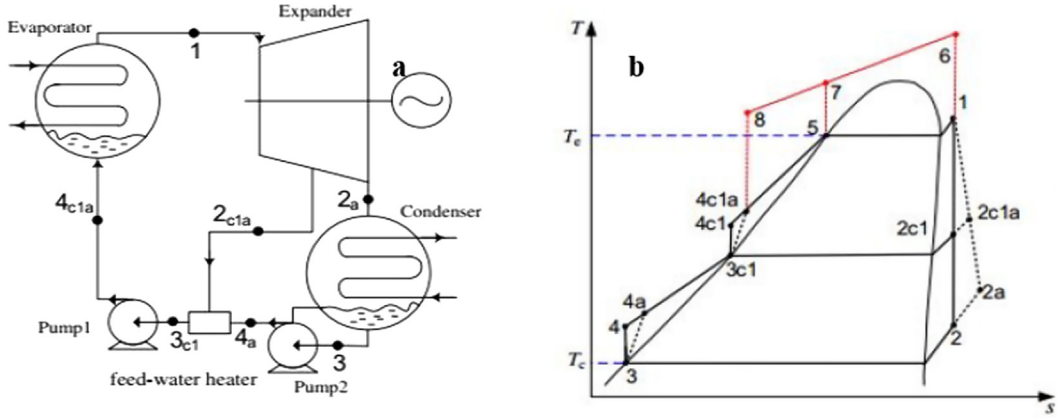

Fig. 3. (a) Schematic of single stage regenerative ORC (SRORC) and (b) T-S diagram for SRORC [42].

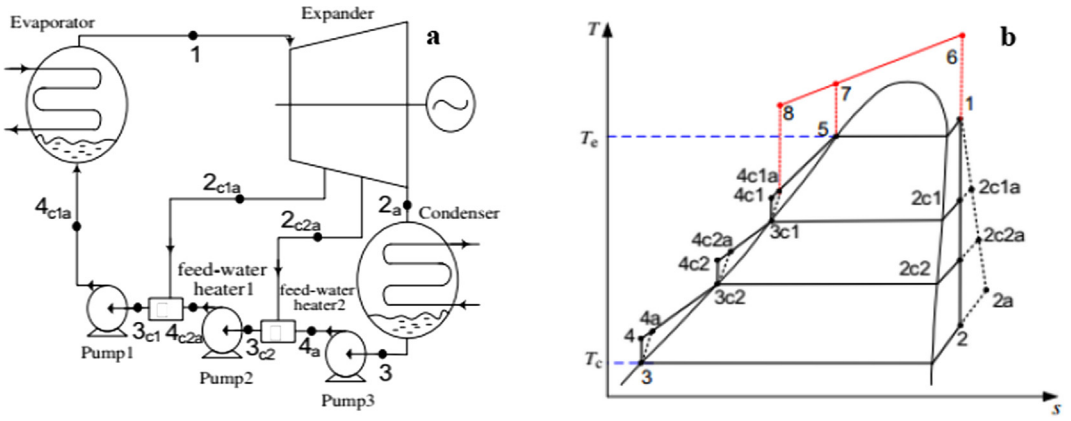

Fig. 4. (a) Schematic of double stage regenerative ORC (DRORC) and (b) T-S diagram for DRORC [42].

capacities of the Cascade-ORC.

Huang et al. [70] comparatively studied performance of two different dual-loop ORCs as shown in Fig. 10. These two systems both had high and low temperature loop but their stage number for heat recovery from engine exhaust was different. They found that the cycle with single stage (Fig. 10a) had better performance compared to the cycle with two stages. Besides, Shu et al. [71] presented a dual-loop ORC that included a high temperature (HT) loop to recover the waste heat of the exhaust, and a low temperature (LT) loop to recover engine coolant and residual heat of the HT loop. A conclusion was that the system showed the best performance at high operating load. Since dual loop systems require large space, are more complex, economically unfavorable, Kim et al. [72] presented a single-loop ORC with R134a for waste heat recovery from both exhaust gas (high temperature heat source) and engine coolant (low temperature heat source). The results showed that the SLORC system has the highest output net power compared to the other conventional ORC system at the fixed engine condition (produced about 20% additional power from waste heat recovery when working at the target engine conditions). Motivated by simpler architecture, smaller volume and higher efficiency compared with conventional dual-loop ORC systems, Chen et al. [73] investigated novel cascade ORC system for waste heat recovery of truck diesel engines as shown in Fig. 11. They found that the novel ORC system produced 7.94% more net power in most of the operating conditions in average compared with dual-loop ORC system. Shi et al. [74] presented a design of ORC for waste heat recovery from unsteady heat sources, such as exhaust gas of automobile engines. In this system, the exhausting gas from the evaporator is partially recovered and mixed with the exhausting gas from the engine, and then re-entered to the evaporator improving the waste heat utilization. The results showed improvement in output power and cycle efficiency. Song et al. [75] performed a thermodynamic analysis for waste heat recovery from both the jacket cooling water and the engine exhaust gas of marine diesel engines. The results indicated that the two separated ORC systems with R245fa and benzene improved the efficiency of the engine about 10%. The optimized ORC system with cyclohexane had a better performance compared to the two separated cycles from economic point of view.

Yun et al. [76] investigated performance of a dual parallel ORC with comparison to the single ORC for waste heat recovery from exhaust gas of the marine engine. Dual ORC could generate higher output power compared to the distribution models that can be achieved by a single ORC. Also, Sung et al. [77] investigated performance of a dual-loop ORC for waste heat recovery as shown in Fig. 12 from dual-fuel engine. For the HT-ORC, generated net output work by ORC with a preheater was higher than the ORC with a recuperator. Yu et al. [78] investigated performance of an ORC in which intermedium hot water had been used. The performance was compared with direct integration system (without intermedium hot water). Compared to the standalone ORC system, the net power output was raised about 24.3% for the system with intermedium hot water.

In another attempt on two-stage configuration, Soffiato et al. [79] investigated the performance of a simple, regenerative two-stage ORC for waste heat recovery from the jacket water, lubricating oil and charge air cooling of diesel engines at 85% load (23,375 kW total power). Two-stage ORC had the highest net power output (about 820 kW) which was approximately twice the simple and regenerative cycle (430–580 kW). Shu et al [80] constructed and tested an Oil Storage/ORC for waste heat recovery from exhaust gas of a 240 kW diesel engine (see Fig. 13). Due to the inertia of the thermal oil, the OS/ORC system could correctly operate when the engine condition changed vastly. Yu et al. [81] experimentally investigated waste heat recovery from the exhaust gas of a heavy-duty diesel engine by using a cascaded

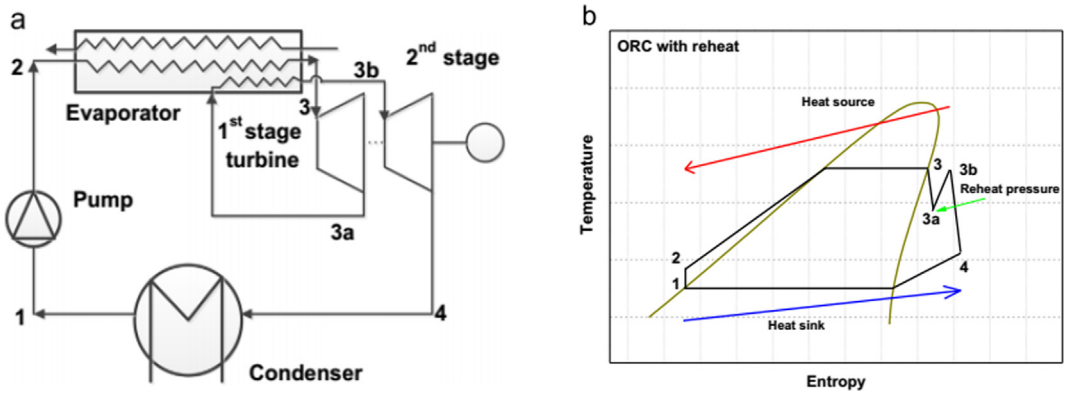

Fig. 5. (a) Schematic of reheat ORC (RORC) and (b) T-S diagram for RORC [44].

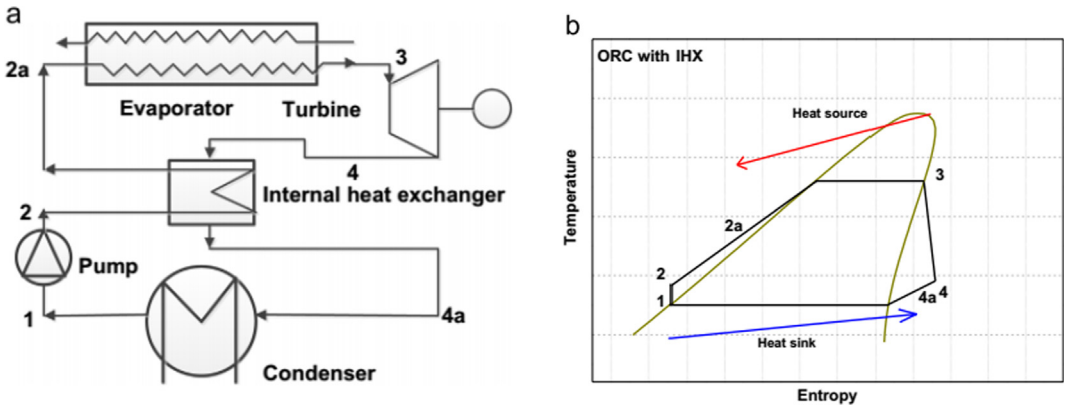

Fig. 6. (a) Schematic ORC with recuperator and (b) T-S diagram for ORC with recuperator [44].

Steam/Organic-Rankine cycle as shown in Fig. 14. It was shown that the combined DE&RC/ORC system had the best performance with up to 5.6% and 3.2% power increment comparing to DE and the combined DE&OS/ORC, respectively. Uusitalo et al. [82] performed experiments to evaluate the performance of small scale ORC for waste heat recovery from exhaust gases of 100–200 kW diesel engine. The thermal energy in the exhaust gas was efficiently recovered by the ORC system. The maximum potential to generate output power with the proposed system was about 9.8 kW.

The summarized data of the studies on the effects of configuration have been presented as the form of stack bar and table in Fig. 15 and Table 1. As it is depicted, for waste heat recovery from internal combustion engines, diesel engines and gas turbines which are the three most favorites heat sources; regenerative single loop, single loop, and regenerative dual loop ORCs have been the most studied thermodynamics cycles, respectively.

3.2. Effect of working fluids on waste heat recovery

Generally, working fluid with lower critical temperature had a higher evaporation temperature. By increment of evaporation temperature, internal exergy efficiency increased and external exergy efficiency showed parabolic-like curves. As a result, working fluid with lower critical temperature could lead to higher overall exergy efficiency [83]. Zeotropic mixtures may be used as working fluid to conquer the siloxanes deficiency such as flammability and great temperature glide that reduce the net output power. For instance, for a DORC, D4/R123 (0.3/0.7) had the best thermodynamic performance with the net power of 21.66 kW, the thermal efficiency of 22.84%, exergy efficiency of 48.6% and the total exergy destruction of 19.64 kW. Economically, MD2M/R123 (0.35/0.65) had the best performance with the smallest EPC of 0.603 $/kW h [84]. Furthermore, Zeotropic binary mixtures compared to the pure fluids have better performance for both subcritical and supercritical operation. Supercritical cycle compared to the subcritical cycle has higher exergy efficiency because of the low critical temperatures for medium and high waste heat temperatures. For example, supercritical mixtures of butaneepropane, butaneehexane and butaneecyclopentane showed better performance compared to the other mixtures [85]. In a proposed regenerative supercritical-subcritical DLORC, for environmental friendliness consideration and because of the thermodynamic performance, among the four pairs of working fluids, R1233zd and R1234yf were the most proper working fluids [86]. For a case study for regenerative ORC with exiting gas temperature of 416 °C, benzene or cyclohexane had the best thermodynamic performance, while the economic analysis showed that benzene was better working fluid compared to cyclohexane [87]. For another case used for heat recovery in cement industry, cyclohexane with higher turbine inlet pressure had the highest exergy efficiency and output power. However, from the exergo-environmental point of view, benzene had the best performance [88]. For application of exhaust stack of a general cargo ship with five-cylinder two-stroke main engine, R113 resulted in the highest net power output and lowest exergy destruction. It was the most proper working fluid with lower toxicity and flammability among others [89]. In addition, maximizing the net power output and minimizing the total investment cost for various working fluids for heat recovery from a diesel engine led to R601a and R245fa. However, by overall consideration of thermo-economic performances, environmental impacts, and safety levels, R245fa was the best choice for the engine waste heat recovery [90]. Hærvig et al. [91] used genetic optimization algorithm to maximize the net output work based on optimal combination of turbine inlet pressure and temperature, condenser pressure, hot fluid outlet temperature, and mixture composition. Optimum working fluid for maximum net output work had critical temperature about 30–50 K lower than the hot source temperature. For working fluids with the same critical temperature, the working fluid with a positive slope of vapor saturation line showed better

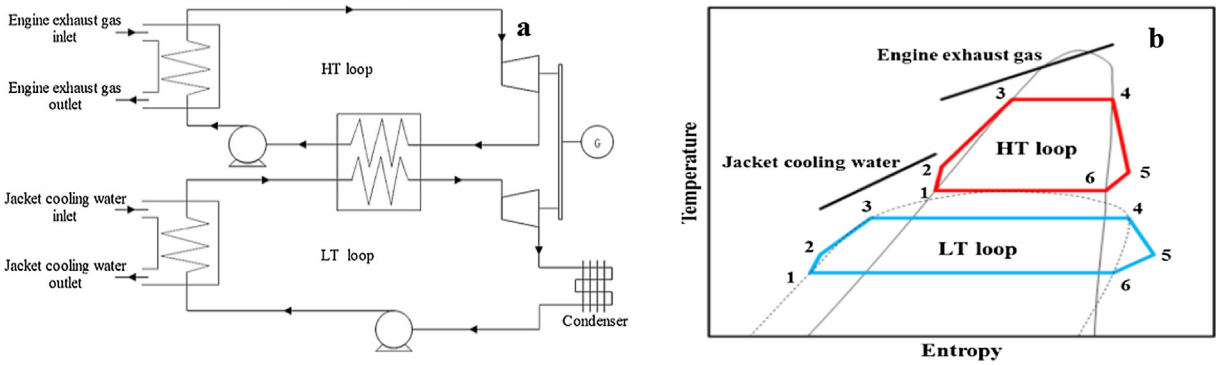

Fig. 7. (a) Schematic of dual loop ORC (DLORC) and (b) T-S diagram for DLORC [50].

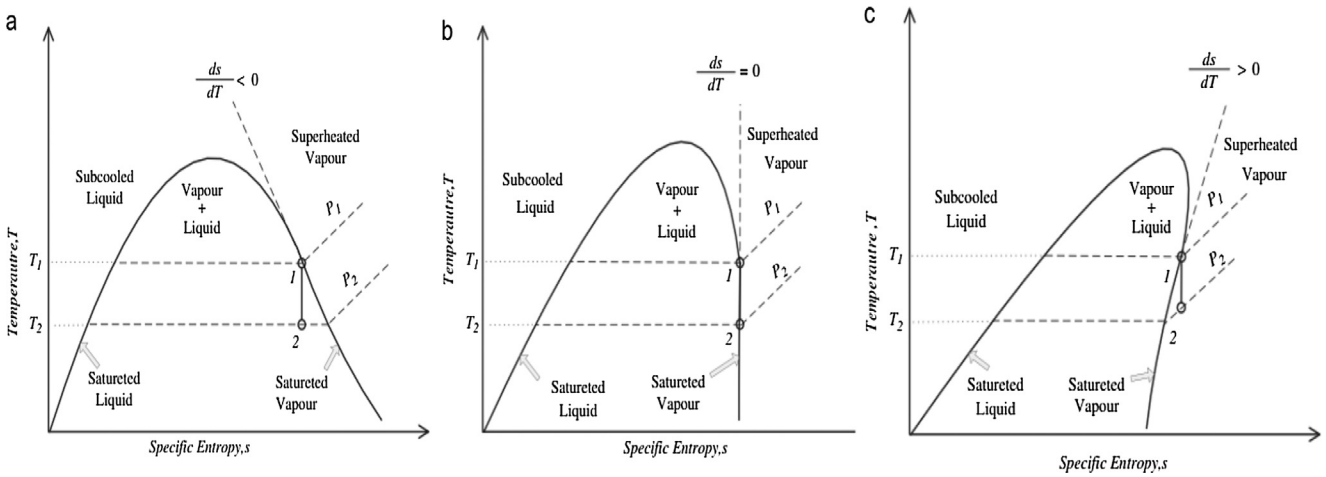

Fig. 8. T–S diagram for (a) wet fluid, (b) isentropic fluid, and (c) dry fluid [64].

performance. In terms of hot source temperature range, a sample of optimal working fluids are listed in Table 2.

Yu et al. [92] performed techno-economic analysis to find best working fluids based on electricity production cost (EPC), depreciated payback period (DPP) and savings-to-investment ratio (SIR). Toluene and R143a had the best economics performance compared to the other working fluids with the lowest EPC (0.27 Dollar/kW h), DPP (7.8 years) and the highest SIR (1.6). In case of using ethanol, a subcritical cycle without recuperator showed best performance economically [93]. Analysis with pure and mixture working fluids based on considering exergy efficiency and levelized energy cost (LEC) as optimization functions has shown that mixtures do not always show better performance compared to the pure fluids and their behavior depend on operating parameters and mass fraction of mixtures [94]. For a boiler in a 240 MW pulverized coal-fired power plant, mixtures which were matched with heat sink and heat source had the highest thermal efficiencies and lower superheat degree, respectively [90]. For a diesel engine heat recovery, Methanol showed the best performance compared to the two other working fluids and it was more stable than toluene, although the cycle with toluene had the highest net power [95]. Song et al. [96] investigated thermodynamics performance of dual loop ORC in which a HT loop was used for heat recovery from the engine exhaust gas, and a LT loop was used to heat recovery of the jacket cooling water and the residual heat of the HT loop. Water was used as the working fluid for the HT loop, while R123, R236fa and R245fa were used in the LT loop. The HT loop with a lower dryness fraction of the wet steam at the inlet of the expander showed better performance, and the wet steam with a dryness fraction of 0.2 generated the maximum net power. Carcasci et al. [97] investigated performance of an ORC with and without superheater under four different working fluids; toluene, benzene, cyclopentane and cyclohexane for waste heat recovery from GE10-1 gas turbines. Cyclohexane, benzene and toluene showed the best performance for low, medium and high oil temperature, respectively. For the application of waste heat recovery from hot water source, with a temperatures range of 100 °C - 150 °C, for minimization of the defect of efficiency the working fluids with low values of critical temperature, such as Novec649, RE347mcc, and R245fa showed the best performance [98]. Li et al. [99] performed thermo-economic analysis of an ORC by using zeotropic mixtures with different components and composition proportions for waste heat recovery from flue gas of an industrial boiler. The mixture has a temperature glide, as the phase change occurs in a temperature range rather than at a constant temperature. The temperature glide on temperature-composition diagram, is defined as the temperature difference between the bubble point and dew point. They found that mixture composition had crucial effect system performance through the temperature glide during the phase change of the mixture. Furthermore, a comparative economic analysis was implemented for steam and Alkane-based Rankine cycle showing a better performance for waste heat recovery using Alkane-based ORC [100]. Hydrocarbons are generally flammable that limits their practical applications and using mixtures of a hydrocarbon and a retardant could conquer this defect. In another case study on a diesel engine, using cyclohexane/R141b (0.5/0.5) led to the best performance for the heat recovery by generating a net output power of 88.7 kW, which increased

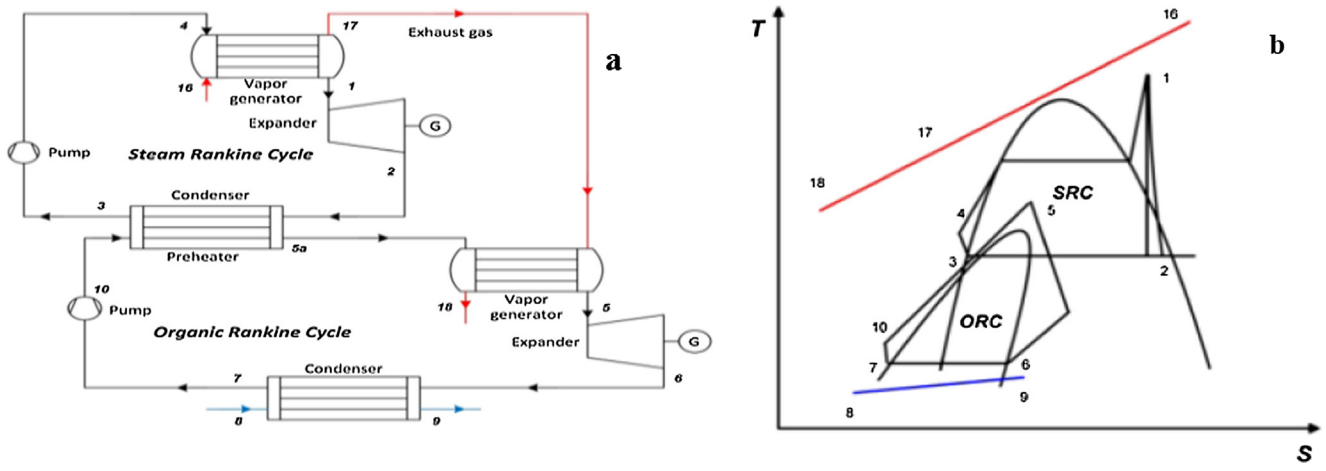

Fig. 9. (a) Schematic of the combined steam-ORC and (b) T-S diagram for the combined steam-ORC [68].

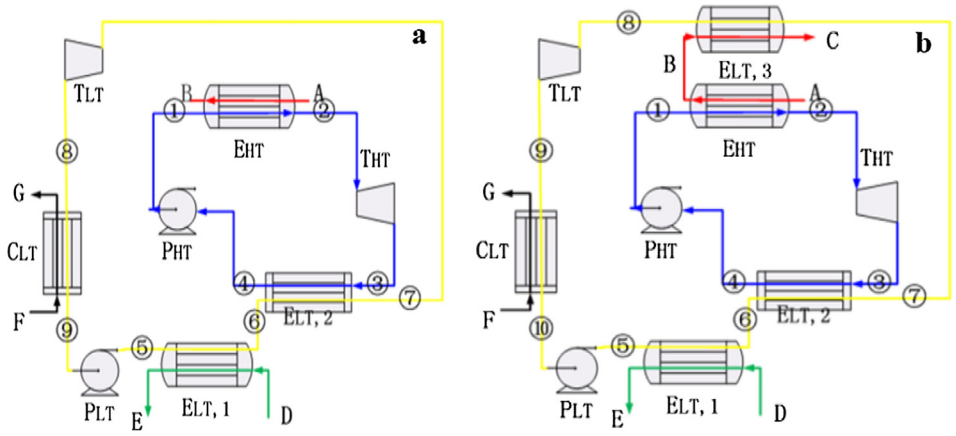

Fig. 10. (a) The first dual-loop ORC and (b) the second dual-loop ORC [70].

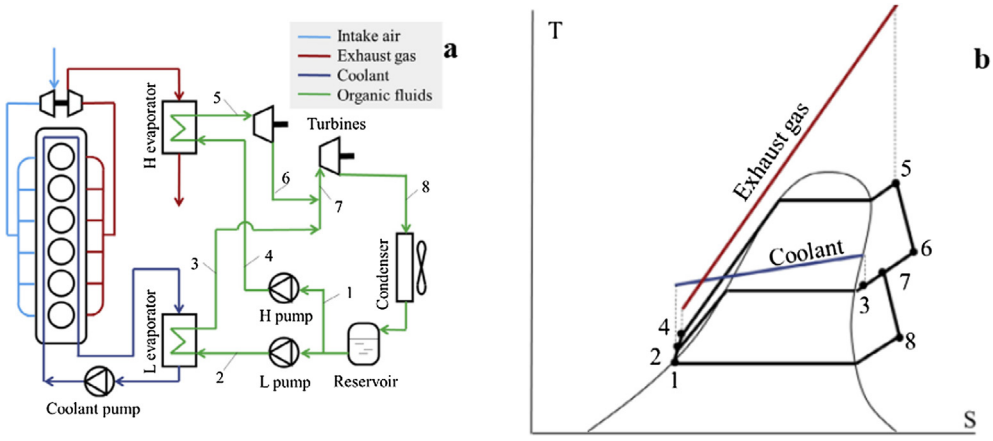

Fig. 11. (a) Schematic of the expansion-ORC and (b) T-S diagram for expansion-ORC system [73].

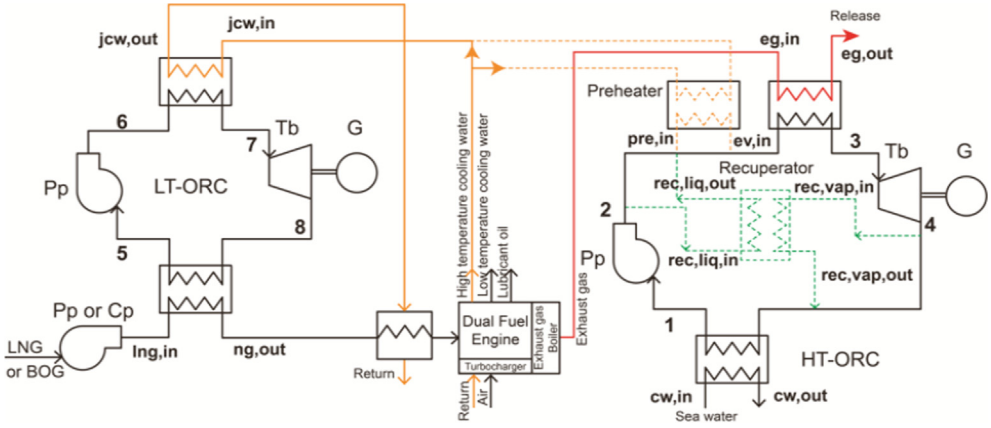

Fig. 12. Schematic diagram of DL-ORC system [77].

the net output power of the system by 13.3% compared to pure cyclohexane [101]. Also, a study on mixtures included pure hydrocarbons: cyclopentane, cyclohexane, benzene and two retardants R11, R123 at various retardants mass fraction and evaporation temperature has shown that an optimum ratio exists for having the best performance for each mixture [102]. For internal combustion engines, the engine does not always operate at a specific load. The pinch point temperature difference is changing stochastically, while the sink temperature is affected by the environment temperature. In a study, hydrocarbon working fluid led to higher net output power compared to the refrigerants because of higher critical temperature leading to higher evaporation temperature [103]. For waste heat recovery from the jacket water of large marine diesel engines, R245ca, R245fa, and R1234yf had higher thermal efficiency compared to the other working fluids. R600a was the optimal working fluid by considering the ratio of the net power output to the total heat-exchanger area [104]. Likewise, Yang et al. [105] performed thermodynamic and economic analysis on an ORC system for waste heat recovery from exhaust gas of large marine diesel engine of the merchant ship. R245fa and R1234ze had the highest net power output and thermal efficiency, respectively. Furthermore, comparison between the cases with and without pre-heater showed that the use of pre-heater improved the net power output and

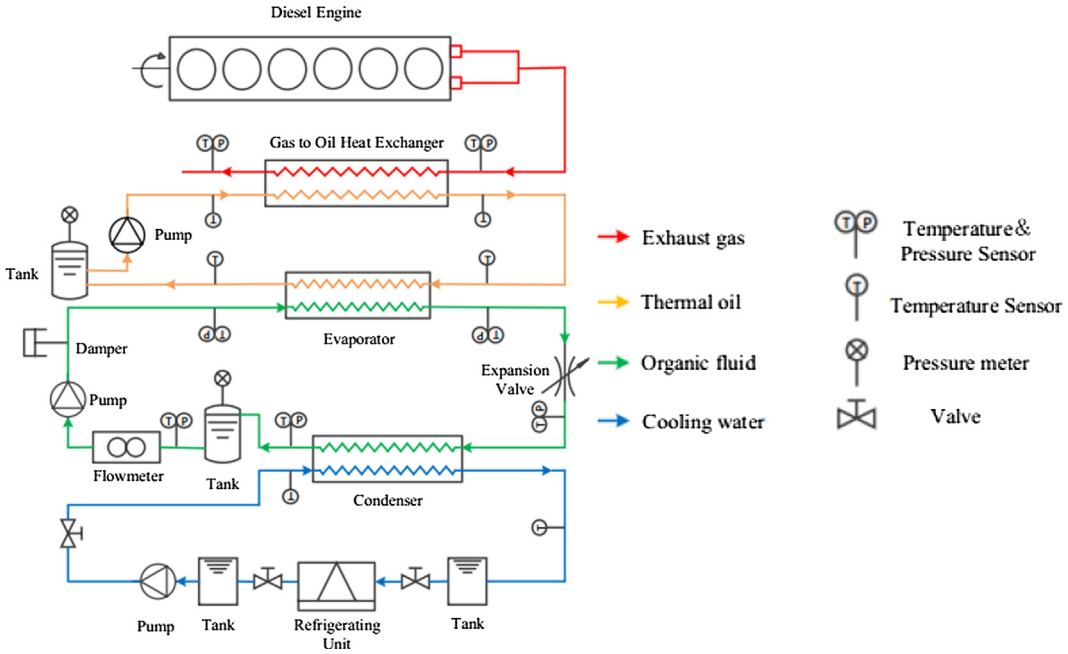

Fig. 13. Structure of the OS/ORC system [80].

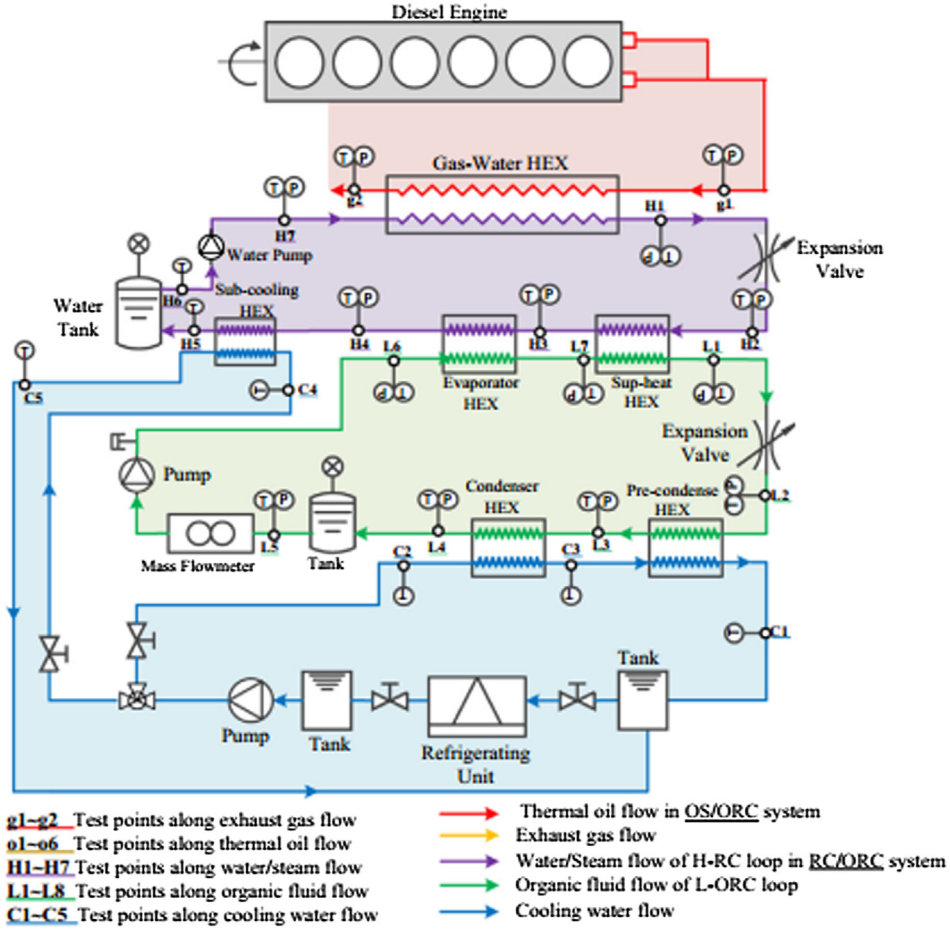

Fig. 14. RC/ORC system test bench [81].

the thermal efficiency. The largest effect of pre-heater was seen when R1234ze was used as working fluid. Generally, the second law efficiency of an ORC with recuperator is higher than the simple cycle. For the simple cycle, the second law efficiency is heavily dependent on the selection of the working fluid. However, for the ORC with recuperator, it is reported to be relatively independent [106]. The effect of working fluid is also investigated in a combined compressed air energy storage (CAES) and ORC system. As the operating pressure increased, the power

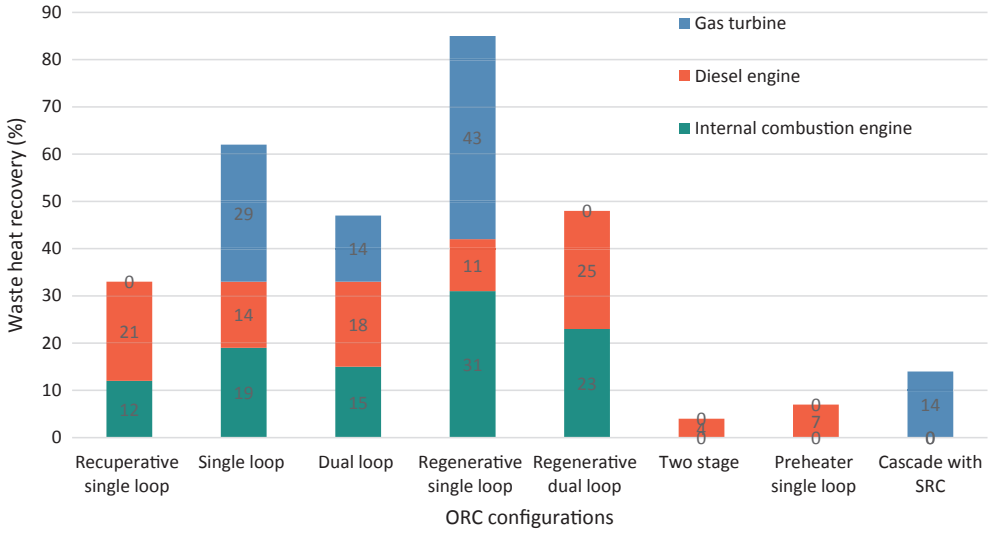

Fig. 15. Statistics on ORC configurations.

Table 1
Summarized data on effects of ORC configuration.

ORC configurations	Working fluids	Waste heat recoveries/ efficiencies	References
Subcritical RC combined with transcritical ORC	R124	23.79%	[68]
Transcritical ORC	R124	21.96%	
Transcritical cascade-ORC with HT and LT loops	Toluene and R143a	9.9%	[69]
Single stage DLORC with HT and LT loops	Water and R143a	14.13%	[70]
Two stages DLORC with HT and LT loops	Water and R143a	14.02%	
DLORC with HT and LT loops	Water and R600	20.07%	[71]
SLORC with preheater	R134a	34.5%	[72]
SLORC with vaporizer	R134a	30.2%	
Novel SLORC	R134a	36.7%	
CCE-ORC	Cyclopentane	11.67%	[73]
DLORC	Cyclopentane	11.39%	
EGMR-ORC	R134a	9.6%	[74]
ORC	R134a	7.1%	
Optimized ORC	Cyclohexane	21.2%	[75]
Two separated ORC	R245fa and benzene	17.4%	
Dual parallel ORC	R245fa	9.4%	[76]
Single ORC	R245fa	10.1%	
DLORC	n-pentane and R125	19.62%	[77]
DLORC with recuperator	n-pentane and R125	21.4%	
DLORC with preheater	n-pentane and R125	22.01	
DLORC with recuperator and preheater	n-pentane and R125	22.5%	
ORC with intermedium hot water	R600	10.4%	[78]
ORC without intermedium hot water	R600	8.7%	
Two-stage ORC	R-236fa	8.39%	[79]
Simple ORC	R-245ca	7.39%	
Cascaded OS/ORC	R123	14.8%	[80]
Cascaded RC/ORC	Water	16.4%	[81]
Cascaded OS/ORC	Thermal oil	13.7%	
ORC with recuperator	siloxane MDM	21.04%	[82]

output and operating temperature of working fluids increased [107]. Using six hydrocarbons and two hydrofluorocarbons, an optimization of twenty-eight possible binary combinations at heat source temperatures ranging from 80 to 180 °C has shown that there is no specific single mixture satisfying all the objective functions together. At a high heat source temperature (180 °C), R245fa and propane with mixtures of 50% showed the best performance, while at a low temperature (80 °C), most of the working fluids showed the same performance [108]. In terms of pinch point design and optimization, both evaporator and condenser pressures could be optimized simultaneously by optimization of working fluid mass flow rate in order to obtain maximum net output work or heat recovery efficiency. In a study, ammonia was the best choice at lower mass flow rates because of lower turbine staging and turbine size and higher exergetic efficiency [109]. An experimental investigation is performed on performance of a regenerative ORC containing a single screw expander for waste heat recovery from a low grade thermal energy source with temperatures up to 125 °C. For a fixed rotational speed, the expander output power of the cycle with R245fa was higher compared to the SES36 [110]. After optimizing of heat exchangers, in an experimental attempt, two different organic fluids (ammonia and HFC-134a) were used for waste heat recovery from exhaust of a diesel engine. The results showed that about 10%, 9% and 8% additional power could be obtained with the optimized shell and tube heat exchanger by using water, ammonia and HFC-134a as working fluids, respectively [111]. Another experimental study, focused on the performance of an ORC system, included a radial-inflow turbine for waste heat recovery from the exhaust gases of a truck combustion engine. Based on all the obtained results, R1233zd was a better choice compared to R245fa for waste heat recovery from the exhaust gases of a long haul truck [112]. Pu et al. [113] investigated a small scale ORC for waste heat recovery from low temperature water. The cycle included R245fa and HFE7100 as working fluids and a single stage axial turbine expander coupled with a permanent magnet synchronous generator. The maximum electric output generated by the turbine expander was obtained by the system with R245fa which was 1979 W, while the maximum electric power output for the HFE7100 working fluid was 1027 W. In a four stroke heavy-duty diesel engine, under various experimental engine operation conditions, the ORC system with R123 had a maximum fuel consumption improvement about 2.8% [114]. Eyerer et al. [115] experimentally obtained the performance of an ORC with two different working fluids for low temperature heat utilization. The system performance was compared between R1233zd-E and R245fa working fluids. R1233zd-E was selected since it had almost no ozone depletion potential and significantly smaller global warming potential compared to R245fa. The results showed that R1233zd-E could be a proper alternative for R245fa with a remarkable reduced global warming potential. As can be seen in Fig. 16 and Table 3, the

Table 2
The optimal working fluids in terms of net power output in different hot source [91].

Hot source temp. range (°C)	Working fluid
50–60	R23
65–70	Ethane
75–90	R7146
95–120	R218
125–160	R227ea
165–170	R124
175–185	R236ea
190	R245fa
195–200	Ipentane
205–235	Pentane
240–255	R123
260–280	R141b

summarized data of the studies on the effects of working fluids have been shown. It can be followed that the most studied working fluids in the reviewed material have been Toluene and R123 for internal combustion engines. For diesel engines R143a, R123, and R245fa; and for gas turbines R152a have been the most studied working fluids for waste heat recovery.

3.3. Effect of operating condition on waste heat recovery

An experimental investigation of a transcritical organic Rankine cycle included a screw expander and R218 for a low grade waste heat recovery. The experiments were performed at heat source temperatures of 90–100 °C in both subcritical and supercritical states for different working conditions. As a result, the inlet pressure of the expander had a positive effect on working fluid's mass flow rate and net efficiency of the system. For heat source temperatures of 90, 95, and 100 °C, the obtained net thermal efficiencies were 2.75%, 2.66% and 2.63% [116]. In a regenerative ORC system integrated into a passenger vessel for waste heat recovery of the engines exhaust gases over a standard round trip, a quasi-steady state simulation on off-design conditions is performed. The ORC system generated average net power of 395.73 kW over a round trip that represented approximately 22% of the total power consumption on board [117]. For a regenerative ORC three ranges for the flue gas inlet temperatures are considered by defining two characteristic temperatures $T_{w1}(1)$(478.65 K) and $T_{w1}(2)$(487.55 K) for the flue gas inlet temperature. When the inlet temperature was lower than$T_{w1}(1)$, the regenerator should not be equipped, the optimal pinch point temperature difference was the lowest limit value, and the optimal evaporation temperature increased by the flue gas inlet temperature. When the inlet temperature was between $T_{w1}(1)$and $T_{w1}(2)$, the regenerator should be equipped, the optimal pinch point temperature difference was the lowest limit value, and the optimal evaporation temperature was the highest limit value. Finally, when the inlet temperature was higher than $T_{w1}(2)$, the regenerator should be equipped, the optimal pinch point temperature difference increased with the inlet temperature, and the optimal evaporation temperature was the highest limit value [118]. A parametric analysis is performed to investigate the effect of operating parameters including evaporator outlet temperature, evaporator outlet pressure, condenser temperature, degree of superheat and pinch point temperature difference on the system performance. Among the five parameters, evaporator outlet temperature had the most important effect on cycle efficiency of the system [119]. Yagli et al. [120] performed a parametric optimization and exergetic analysis in order to compare the performance of subcritical and supercritical ORC for waste heat recovery from exhaust of a biogas fueled combined heat and power (CHP) system by using R245fa and under various turbine inlet temperatures and pressures. The subcritical and supercritical ORC have respectively lower and higher turbine inlet pressure than that of critical pressure. In a transcritical ORC for low-grade waste heat recovery, the effect of different operational parameters including mass fractions of R1234yf/R32 mixtures, isentropic efficiencies of the expander, condensation temperature, turbine inlet pressure and temperature were studied on the system performance. The optimal mass fraction of R32 increased with the expander inlet pressure and temperature and decreased with the condensation temperature [121]. Moreover, in a marine heavy duty diesel engine, a two-stage ORC is integrated with a reverse osmosis desalination (RO) unit for a cogeneration. Design parameters were evaporator's pressure, seawater salinity and volumetric flow rate of the fresh water. Increment of the first evaporator pressure, volume flow rate of fresh water, and seawater salinity had negative effect on the net output power and exergy efficiency [122]. A parametric study was performed on performance of an ORC combined with Absorption Refrigeration Cycle (ARC) and the Ejector Refrigeration Cycle for waste heat recovery from an industrial low-temperature heat source. R113 was used as the working fluid and the effects of various parameters including evaporation temperature, condensation temperature, and degree of superheat were investigated

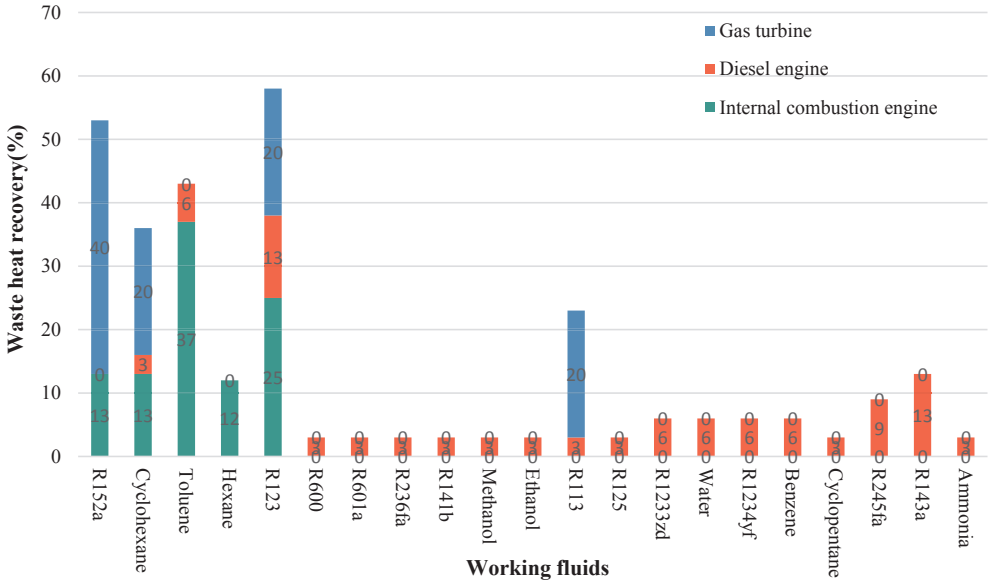

Fig. 16. Statistics on working fluids.

Table 3
Summarized data on effects of working fluids.

Working fluids	ORC configurations	Waste heat recoveries/exergy or thermal efficiencies	References
Butane, R123, R245fa, R600a, R245ca, R141b	Basic ORC	25.1%, 23.8%, 25.4%, 26.1%, 24.2%, 23.4%	[83]
D4/R123, MDM/R123, MD2M/R123	DLORC	22.84%, 20.41%, 20.02%	[84]
Butanee/Cyclopentane(6/4), ButaneePropane(3/7)	Subcritical ORC	15.53%, 17.53%,	[85]
Butanee/Propane(6/4), Butanee/Hexane (9/1), Butanee/Cyclopentane(3/7)	Supercritical ORC	30.5%, 29.05%, 39.77%	
R1233zd, R245fa, toluene, water, R1234yf, R134a, R143a, ethanol	Regenerative supercritical-subcritical DLORC	21.16%, 20.65%, 21.42%, 23.45%, 10.9%, 10.8%, 10%, 11.25%	[86]
Benzene, cyclohexane, Toluene, P-xylene	Regenerative ORC	19.4%, 19.1%, 18.8%, 18.3%	[87]
Benzene, cyclohexane, Toluene, P-xylene	Basic ORC	16.1%, 17.4, 16.3%, 15.9%	
Cyclohexane, benzene, toluene	ORC with preheater	27.11%, 25.62%, 18.04%	[88]
R113, R245FA, R601, R601A	ORC with preheater	19.69%, 19.42%, 19.39%, 19.46%	[89]
R290, R600, R600a, R601, R601a, R134a,		10.62%, 11.33%, 11.28%, 11.34%,	[90]
R227ea, R245fa	Basic ORC	11.17%, 10.54%, 10.16%, 11.36%	
Toluene, R143a, decane	Transcritical cascade-Organic	16.41%, 15.7%, 13.84%	[92]
Pentane, R245fa, Ethanol	recuperative subcritical ORC	12.4%, 13.7%, 12.61%	[93]
Pentane, R245fa, Ethanol	recuperative supercritical ORC	16.36%, 14.81%, 14.22%	
Pentane, R245fa, Ethanol	supercritical ORC	14.2%, 13.95%, 15.48%	
Pentane, R245fa, Ethanol	subcritical ORC	13.7%, 12.6%, 14.05%	
R245fa/R227ea(9/1), R245fa/R227ea(8/2), R245fa/R227ea(7/3), R245fa/R227ea(6/4)	Basic ORC	49.42%, 49.4%, 48.86%, 47.41%	[94]
methanol, toluene, Solkatherm SES36	ORC with preheater	24.1%, 20.8%, 7.86%	[95]
R123, R236fa, R245fa	DLORC	8.1%, 10.9%, 8.92%	[96]
R365mfc (HST = 100 C)	Basic ORC	18%	[98]
Novec649 (HST = 150 C)	Basic ORC	28%	
R245fa, R245fa/R152a(9/1), R245fa/R152a (6.5/3.5), R245fa/R152a(4.5/5.5)	Basic ORC	10.46%, 8.7%, 8.48%, 9.1%	[99]
Cyclohexane, Decane, Nonane, Octane, Heptane, Cyclopentane, Hexane, Isohexane	Basic ORC	18.02%, 17.64%, 17.57%, 17.12%, 16.8%, 16.64%, 14.4%, 13.62%,	[100]
Cyclohexane, Benzene, Toluene, Cyclohexane/R141b(5/5), Cyclohexane/R11(5/5)	ORC with preheater	18%, 17.4%, 17.31%, 18.6%, 18.3%	[101]
Cyclohexane/R123(7/3), Benzene/R123(7/3), Cyclohexane/R11(7/3), Benzene/R11(7/3)	ORC with IHE	15.94%, 16.58%, 16.1%, 16.7%	
Cyclohexane/R123(7/3), Benzene/R123(7/3), Cyclohexane/R11(7/3), Benzene/R11(7/3)	ORC without IHE	1.73%, 15.03%, 13.97%, 15.23%	[102]
Benzene, toluene, isohexane, hexane, pentane	Basic ORC	14.1%, 13.02%, 11.92%, 11.91%, 11.04%	[103]
R245fa, R600a, R245ca, R1234yf, R1233zd	ORC with preheater	4.48%, 4.38%, 4.56%, 4.08%, 4.46%	[104]
R1234ze, R245fa, R600, R600a	ORC with preheater	10.51%, 10.08%, 10.21%, 10.37%	[105]
R1234ze, R245fa, R600, R600a	ORC without preheater	9.61%, 9.46%, 9.5%, 9.42%	
R123, R134a, R152a, R245fa, R600a	ORC with recuperator	8.42%, 7.1%, 7.8%, 8.12%, 7.83%	[107]
pentane, isopentane, neopentane, R245fa, butane, isobutane, R134a	Subcritical ORC	11.12%, 10.94%, 10.67%, 10.4%, 10.32%, 10.2%, 10.09%	[108]
Isopentane, Ammonia, Benzene, Toluene, n-Pentane, R-1233zd	subcritical-supercritical ORC	18.01%, 17.6%, 13.4%, 13.61%, 17.83%, 17.92%	[109]
SES36, R245fa	ORC with recuperator	9%, 7.7%	[110]
Ammonia, HFC-134a, water	ORC with super heater	10.03%, 9.42%, 11.46%	[111]
R245fa, HFE7100	Basic ORC	7.9% 7.2%	[113]
R123, R245fa	Basic ORC	9.72%, 8.83%	[114]
R1233zd-E, R245fa	Basic ORC	11.3%, 10.0%	[115]

on the thermodynamic performance of the cycle. There was a maximum exergetic efficiency as the evaporation temperature increased [123]. Yang et al. [124] performed a parametric study on a dual loop ORC system that is used for heat recovery from intercooler heat rejection of a six-cylinder CNG engine under various operating range. The optimum evaporation pressure and superheat degree range for the HT cycle under different operating conditions were 2.5–2.9 MPa and 0.43–12.35 K, respectively. Furthermore, an investigation is performed for a double loop ORC for waste heat recovery of a natural gas engine with 1000 kW rated power under five typical engine working conditions (60, 70, 80, 90 and 100%) and various mass flow rates of the high temperature cooling water. The results indicated that DORC could improve the efficiency of the combined system above 60% engine working condition [125]. A parametric study is conducted on thermodynamic performance of a combined ORC and PEM electrolysis to generate hydrogen through waste heat recovery from a gas turbine modular helium. The turbine inlet temperature had a positive effect on the exergy efficiency and the rate of produced hydrogen which improved the system performance. Also, the exergy efficiency and the rate of produced hydrogen had an optimum value with the evaporator temperature [126]. Another parametric analysis is performed on a dual-loop ORC system for waste heat recovery from a light-duty diesel engine. The expander isentropic efficiency and the evaporation pressure in the HT loop increased the net power output of the system, and the condensation temperature of the LT loop had a negative effect on the system performance [127]. Beside theoretical analysis, experiments are also conducted for exploring the effects of operating conditions. An experimental study on waste heat recovery from a diesel engine exhaust with an ORC is performed by using a single-screw expander. The single-screw expander worked properly for small/medium scale ORC system at low/medium rotational speed with a maximum output power [128]. Low-grade waste heat recovery from steam with pressure range of 1–3 bar by using an ORC was experimentally inspected. With increment of superheating by 1 °C, the thermal efficiency decreased about 0.021%; therefore, the best operating condition was nearly no superheating [129]. In a ceramic industry in which heat was transmitted by a thermal oil from the heat source to the ORC, the measurements showed that the thermal oil temperature had a positive effect on the produced electrical power [130]. Li et al. [131] experimentally investigated the performance of an ORC for waste heat recovery from low grade heat sources

Table 4
Summarized data on effects of operation conditions.

Operation conditions	Working fluids	Configurations	Effect of parameters on cycle efficiency	References
Increamnet of heat source temperatures	R218	Transcritical ORC	Negative effect	[116]
Increment of inlet pressure of the expander	R218	Transcritical ORC	Positive effect	
Increment of heat source temperatures	Benzene	Regenerative ORC	Positive effect	[117]
Increment of heat source mass flow rate	Benzene	Regenerative ORC	Positive effect	
Increment of evaporator outlet temperature	R123	Regenerative ORC	Positive effect	[118]
Increment of evaporator outlet pressure	R123	Regenerative ORC	Positive effect	
Increment of condenser temperature	R123	Regenerative ORC	Negative effect	
Increment of degree of superheated	R123	Regenerative ORC	Positive effect	
Increment of pinch point temperature difference	R123	Regenerative ORC	No effect	
Increment of turbine inlet temperature	R245fa	Subcritical ORC	Positive effect	[119]
Increment of turbine of inlet pressure (4–12 bar)	R245fa	Subcritical ORC	Negative effect	
Increment of turbine of inlet pressure (12–30 bar)	R245fa	Subcritical ORC	Positive effect	
Increment of turbine inlet temperature	R245fa	Supercritical ORC	Positive effect	
Increment of turbine of inlet pressure	R245fa	Supercritical ORC	Positive effect	
Increment of the first evaporator pressure	R245fa/cyclohexane (0.5/0.5)	Two-stage ORC integrated with reverse osmosis desalination	Negative effect	[120]
Increment of the second evaporator pressure			Positive effect	
Increment of evaporation temperatures (86–110 °C)	R113	ORC combined with ARC	Positive effect	[121]
Increment of evaporation temperatures (110–156 °C)	R113	ORC combined with ARC	Negative effect	
Increment of condensation temperatures	R113	ORC combined with ARC	Negative effect	
Increment of degrees of superheat	R113	ORC combined with ARC	Negative effect	
Increment of evaporator pressure of HT cycle	R245fa	DLORC	Positive effect	[122]
Increment of condensation temperature of HT cycle	R245fa	DLORC	Negative effect	
Increment of condensation temperature of LT cycle	R245fa	DLORC	Negative effect	
Increment of evaporator temperature of LT cycle	R245fa	DLORC	Positive effect	
Increment of superheat degree of HT cycle	R245fa	DLORC	No effect	
Increment of engine working load	Toluene for HTL, R245fa for LTL	DLORC	Positive effect	[123]
Increment of mass flow rate of the HT cooling water		DLORC	Positive effect	
Increment of turbine inlet temperature		ORC combined with PEM	Positive effect	[124]
Increment of expander isentropic efficiency	R245fa for HTL, R134a for LTL	DLORC	Positive effect	[125]
Increment of evaporation pressure in the HTL		DLORC	Positive effect	
Increment of condensation temperature in the LTL		DLORC	Negative effect	
Increment of evaporation pressure in the HTL	R245fa for HTL, R134a for LTL	DLORC	Positive effect	[126]
Increment of engine working load		DLORC	Positive effect	
Increment of condensation temperature in the LTL		DLORC	Negative effect	
Increment of heat source temperature	R245fa	Basic ORC	Positive effect	[127]
Increment of ORC pump speeds	R245fa	Basic ORC	Negative effect	
Increment of pressure drop	R245fa	Basic ORC	Positive effect	[128]
Increment of degree of superheating	R245fa	Basic ORC	Positive effect	
Increment of condenser temperature	R245fa	Basic ORC	Negative effect	
Increment of evaporation pressure	R123	Basic ORC	Positive effect	
Increment of heat source temperature	R123	Basic ORC	Positive effect	[129]
Increment of degree of superheating	R123	Basic ORC	Negative effect	
Increment of condenser cooling water temperatures	R245fa	Basic ORC	Negative effect	[130]
Increment of degree of superheating	R245fa	Basic ORC	Negative effect	
Increment of condenser cooling water temperatures	HFC245fa	ORC with recuperator	Negative effect	[131]
Increment of heat source temperature	HFC245fa	ORC with recuperator	Positive effect	
Increment of working fluid mass flow rate	R123	Recuperative ORC	Positive effect	[132]
Increment of pressure ratio	R123	Recuperative ORC	Positive effect	
Increment of condensation temperature	R134a	SLORC	Negative effect	[133]
Increment of vaporization temperature	R134a	SLORC	Positive effect	
Increment of pre-heating temperature	R134a	Basic ORC	Positive effect	[134]
Increment of turbine inlet temperature	R134a	Basic ORC	Negative effect	
Increment of evaporating pressure	Cyclohexane	Inter-heating ORC	Positive effect	[135]
Increment of intermediate pressure	Cyclohexane	Inter-heating ORC	Negative effect	

Table 5
Summarized data on effects of different cycles.

Cycles and Configurations	Working fluids	Heat source	Waste heat recoveries/exergy or cycle efficiencies	References
ORC with recuperator, KC recuperator	R601, Ammonia/water	Gas Turbine-Modular Helium	51.06%, 52.7%	[136]
ORC, KC	R290, Ammonia/water	CGAM cogeneration system	52.74%, 52.49%	[137]
ORC, KC	R113, Ammonia/water	solid oxide fuel cell	62.35%, 61.32%	[138]
ORC, RC, KC	R245ca, water, Ammonia/water	diesel engine	52%, 51.1%, 51%	[139]
HT-ORC, LT-ORC, DLORC, RC	Toluene, R245fa, Toluene/R245fa, water	gas engine	18.2%, 9.12%, 23.7%, 21.28%	[140]
DLORC, RC, BC	R123/R245fa, Water, CO2	internal combustion engine	26.55%, 18.1%, 23.8%	[141]
TLC, KC, ORC	Ammonia/water, Ammonia/water, R141b	low-grade heat source	8.08%, 3.09%, 10.06%	[142]
transcritical ORC, KC	n-Nonane, Ammonia/water	internal combustion engine	31.7%, 18.8%	[143]

to generate electric power at various working conditions. At fixed ORC pump speeds, increment of heat source temperature had a positive effect on the cycle performance, increasing turbine power output, overall efficiency and thermal efficiency. Yang et al. [132] studied the effects of pressure drop, degree of superheating and condenser temperature on performance of the ORC experimentally. The cycle included an open-drive scroll type expander with R245fa as the working fluid and the system was used for low grade waste heat recovery. The pressure drop and degree of superheating had positive effects on the thermal efficiency. In another study, a scroll expander was used with fin tubes heat exchanger as the evaporator for waste heat recovery from industrial flue gas with a temperature range of 90–220 °C. Experiments were performed for various evaporating pressures, heat source temperatures and superheat degrees of the working fluid. The evaporating pressure and heat source temperature both increased the expander output power and the exergetic efficiency [133]. Li et al. [134] experimentally investigated the performance of an ORC with R245fa, at various condenser cooling water temperatures and superheating for waste heat recovery from a low-grade heat source. At constant heat source temperature and flow rate, as cooling water temperatures increased, the turbo-expander power output and cycle efficiency decreased. Minea et al. [135] measured the performance of a laboratory beta-prototype, 50 kW ORC for industrial waste heat recovery with heat source temperature range of 85–116 °C and a cooling fluid temperature range of 15–30 °C. Decrement of cooling fluid and waste heat inlet temperatures had improved and decreased the system performance, respectively. The summarized data of the studies on the effects of operation conditions have been presented in Table 4.

4. Comparison of waste heat recovery of organic Rankine cycles (ORCs) to other cycle

Zare et al. [136] performed a comparative thermodynamic analysis and optimization between organic Rankine cycle (ORC) and Kalina cycle (KC) for waste heat recovery from a Gas Turbine-Modular Helium Reactor (GT-MHR). The results demonstrated that the combined GTMHR/ORC had better performance for heat recovery compared to the combined GTMHR/KC based on the higher cycle efficiency and lower high-pressure level. Another advantage of ORC with respect to the KC, was that the working fluid at the ORC turbine exit was superheated vapor while the KC turbine exit was two-phase flow. Similarly, Nemati et al. [137] conducted a comparison between ORC and Kalina cycle (KC) as a bottoming cycle for waste heat recovery from CGAM cogeneration system. Higher energy and exergy efficiencies, lower optimum pressure value which with lower cost levels for materials and sealing, and superheated vapor at the turbine outlet were the advantages of the combined CGAM/ORC compared to the CGAM/KC cycle. A comparison of the performance of Organic Rankine and Kalina cycles for waste heat recovery from solid oxide fuel cell indicated that ORC had better performance compared to the KC with required noticeably lower values of operating pressures which was another advantage of the ORC over the KC [138]. Furthermore, Larsen et al. [139] compared performance of ORC, Kalina cycle and steam Rankine cycle for waste heat recovery from marine two-stroke low speed diesel engine based on the efficiency, the environmental impact, and safety concerns. Kalina cycle had no remarkable advantage compared to the ORC or the steam cycle. However, ORC increased the NOx emissions in exchange, due to the engine tuning requirements. Regarding different temperatures and heat sources, Shu et al. [140] comparatively studied ORC, HT-ORC, LT-ORC, DLORC and RC for waste heat recovery from gas engine of rated power 1000 kW. The results showed that DORC had the highest output power of 162 kW and the order of highest to lowest thermodynamic performance was DORC > RC > HT-ORC > LT-ORC. However, based on system cost and electricity production cost the order of highest to lowest values were RC > HT-ORC > DORC > LT-ORC and LT-ORC > HT-ORC > RC > DORC, respectively. Using ORC as a bottoming cycle, Zhang et al. [141] comparatively studied waste heat recovery from internal combustion engine by steam Rankine cycle (RC), Brayton cycle (BC) and thermoelectric generator (TEG) as topping recovery cycles. They performed thermodynamic and techno-economy analysis by considering net output power, thermal efficiency and the turbine size parameter (SP), and heat transfer capacity (UA) as key indicators at various engine conditions ranging from high to low loads. Based on techno-economy analysis, dual-loop ORC had large values of SP and UA and TEG–ORC showed better performance compared to the other cycles. With a reduction of engine load, both heat transfer capacity (UA) and turbine size parameter (SP) decreased, and at low engine operation loads TEG–ORC system was superior. Again, a comparison with Kalina cycle is performed by Yari et al. [142] with exergo-economic study of trilateral power cycle (TLC) and ORC for waste heat recovery from a low-grade heat source with a temperature of 120 °C. For the ORC, maximum net output power occurred at specific turbine inlet temperature while for the TLC, the maximum power increased with increment of expander inlet temperature. Net output power of the TLC was heavily dependent on expander isentropic efficiency, and TLC with expander isentropic efficiency close to that of conventional turbines could generate more power compared to the two other cycles. Moreover, Yue et al. [143] compared performance of two bottoming cycles including transcritical ORC and a Kalina cycle for waste heat recovery under different internal combustion engine working conditions (20 – 100% load). For engine working conditions with exhaust temperature over 491 K, the transcritical ORC had better performance based on the overall thermal efficiency, low operational pressure and simple components configuration. The optimal exhaust temperature range for the transcritical ORC was 569–618 K. The summarized data of the studies on comparison of different cycles is indicated in Table 5.

5. Conclusion

This paper presented a comprehensive review on recent developments (since 2014–2018) of ORCs that have been used for power

generation by using a waste heat source. It can be concluded that:

- Diesel engines, internal combustion engines and gas turbines are respectively the most used heat sources (in recent academic papers) for waste heat recovery.
- For diesel engines as heat source, because of the multi-grade waste heat recovery, regenerative dual loop ORCs have mostly been used compared to the other cycles, however, for the cases with internal combustion engines and gas turbines, regenerative single loops have been used frequently.
- The most influential thermos-physical characteristics of working fluids were critical state, sensible heat and ratio of vaporization latent heat. Mixing different working fluids was an effective way to improve these thermos-physical properties. Mixture fluids with compared to pure fluids have shown better performance in average but their compositions and mass fractions have heavily changed the cycle performance, and should be optimized in design process of ORC plants.
- Various cycle parameters such as heat source types, operating conditions and temperature levels could heavily affect the cycle performance. It was shown that increment of heat source and condensation temperature had positive and negative effects on ORC performance, respectively. However, other parameters showed different influences on cycle performance at various conditions, and usually a multi-objective optimization is required based on thermodynamic and economic criteria for each cycle in order to determine favorable operational parameters.

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